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Dated 11 May 2004

GB 0308467.0

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OXFORD BIOSIGNALS LIMITED,
Magdalen Centre,
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Incorporated in the United Kingdom,

[ADP No. 08157893002]

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2.	Patent application number *(The Patent Office will fill in this part)*	0308467.0
3.	Full name, address and postcode of the or of each applicant *(underline all surnames)*	Rolls-Royce plc 65 Buckingham Gate LONDON SW1E 6AT
	00003970002 Patents ADP number *(if you know it)*	SECTION 30 (1977 ACT) APPLICATION FILED 29/3/04
	If the applicant is a corporate body, give the country/state of its incorporation	GB
4.	Title of the invention	METHOD AND SYSTEM FOR ANALYSING TACHOMETER AND VIBRATION DATA FROM AN APPARATUS HAVING ONE OR MORE ROTARY COMPONENTS
5.	Name of your agent *(if you have one)* "Address for service" in the United Kingdom to which all correspondence should be sent *(including the postcode)*	MEWBURN ELLIS LLP York House 23 Kingsway London WC2B 6HP MO 10.5.04
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METHOD AND SYSTEM FOR ANALYSING TACHOMETER AND VIBRATION DATA FROM AN APPARATUS HAVING ONE OR MORE ROTARY COMPONENTS

Field of the Invention

This invention relates to a method and system for analysing tachometer and vibration data from an apparatus having one or more rotary components, and to the use of polar diagrams for modelling and/or monitoring the behaviour of such an apparatus.

Background

Apparatuses with rotary components, such as gas turbine engines, are subject to vibratory forces at frequencies which are related to the angular velocity of the respective component and hence engine speed. These frequencies are conventionally known as engine order forcing frequencies, each engine order corresponding to a rotational frequency of a particular component (or a fraction or harmonic of the fundamental frequency) and exerting a corresponding vibratory force on the engine.

The forces may arise because e.g. an engine is out of balance on a particular shaft, stiffness irregularities in engine components, and (significantly in the case of gas turbine engines) aerodynamic interactions between the blades of the engine.

At a given engine speed, a number of these engine orders are generally active and result in corresponding vibration responses in the engine which are measurable e.g. as strains or accelerations. Each vibration response generally has the same frequency as the engine order forcing frequency which generated it. However, the relative phase difference between a vibration response and the corresponding engine order may change as the engine speed varies, and particularly when the

engine order traverses a resonance frequency of the engine. Indeed, merely moving toward or away from such a resonance may cause the phase difference to change.

A conventional approach for determining the phase relationship between the forcing frequency of an engine component (e.g. a shaft) and a vibration response is to fit a dedicated once per revolution tachometer to the component. This tachometer would determine the component rotational position (i.e. phase) and also serves as a trigger for the collection of vibration measurements. The approach is illustrated by Figure 1 which is a flow diagram showing the sequence of data acquisition and analysis events.

The approach is relatively simple in concept, and the synchronisation between the tachometer and the vibration measurements allows the absolute phase difference between the component rotational position and the vibration response to be determined. However, it relies on being able to fit an accurate, robust and dedicated tachometer to the component in question, something which is not always possible for complex components such as the shafts of multi-shaft gas turbine engines. Also the approach precludes deriving simultaneous phase information for other components (e.g. other shafts in a multi-shaft gas turbine), unless the investigator is able to fit further dedicated tachometers which in turn trigger further vibration measurements. Thus, in relation to gas turbine engines, the approach is only generally used for shaft balancing operations, where absolute phase information is needed.

Summary of the Invention

Aspects of the present invention are at least partly based on the realisation that a relative (rather than an absolute)

phase difference between a forcing frequency and a vibration response can provide useful information about the state of an engine.

A first aspect of the invention provides a method of analysing tachometer and vibration response data from an apparatus having one or more rotary components, the method comprising the steps of:

providing vibration response data and tachometer data from the apparatus, the tachometer data being for a rotary component of the apparatus, and the vibration response data and tachometer data being acquired independently of each other;

determining a forcing frequency of the component from the tachometer data and a corresponding vibration response frequency of the apparatus from the vibration response data; and

comparing the forcing and vibration response frequencies to determine the relative phase difference between the frequencies.

Preferably the apparatus is a gas turbine engine. The rotary component may be a turbine drive shaft of the engine.

The tachometer data and vibration response data are independently acquired of each other, by which we preferably mean that the tachometer is not used to trigger acquisition of the vibration response data. This means that, compared with the conventional approach discussed above, more flexible data gathering arrangements can be adopted. For example, if tachometer data is available for each of a plurality of shafts of a multi-shaft gas turbine engine, the corresponding vibration response frequencies can all be derived from the same source of vibration response data. This source may be e.g. a single vibration transducer.

Although the tachometer data can provide the absolute rotary position of the component, this is not essential and the tachometer data can provide merely the relative rotary position of the component instead. For example, the tachometer may measure rotary position using a multi-pole or other non-unique position indicator, or it may measure the rotary position of the output of a set of gears, which are in turn connected, at a known gearing ratio, to the component of interest. Thus the problem of fitting a dedicated tachometer to the component can be avoided.

In preferred embodiments, the forcing and vibration response frequencies are compared in the time domain in order to determine the relative phase difference between the frequencies.

When performing phase comparisons between signals in the frequency domain, a reasonable level of coherence must be achieved within the cross-transform. However, in the previously-mentioned conventional approach for determining the phase relationship between the forcing frequency of a rotary component and a vibration response in which a tachometer triggers the collection of vibration measurements, the tachometer signal typically takes the form of narrow pulses. These pulses do not generally contain sufficient energy to maintain adequate coherence. A possible solution would be to extend the duration of each pulse by increasing its mark-space ratio, but this is only practicable over certain speed ranges of the component. However, by performing the phase measurement in the time domain these difficulties are overcome. A further advantage is that relative phase differences can be measured on a number of independent components (e.g. shafts), together with component orders which can be fractional multiples of the reference component

rotational speed or can be mechanically coupled e.g. via gear trains.

Preferably the amplitude of the vibration response is also determined from the vibration response data. The phase difference between the frequencies and the vibration amplitude may then be plotted on a polar diagram. In general, apparatuses such as gas turbine engines display characteristic behaviours in relative phase difference and/or vibration amplitude as e.g. the engine accelerates, decelerates or cruises. Departures from these behaviours can be indicative of "abnormal" engine behaviour. Polar diagrams allow such departures to be easily identified.

The phase difference, preferably represented on a polar diagram, may be used to diagnose the state of the apparatus or to identify an event occurring to the apparatus e.g. in order to understand apparatus behaviour, validate apparatus models, troubleshoot the apparatus, monitor the "health" of the apparatus, monitor for abnormal events etc.

Related aspects of the invention provide (a) a computer system operatively configured to perform the method of the first aspect, (b) computer readable media carrying computer code for performing the method of the first aspect, and (c) a computer program for performing the method of the first aspect.

Optional and/or preferred features of the first aspect of the invention may also be applied to the related aspects. Thus, for example, the above computer system may be a system for diagnosing the state of the apparatus or for identifying an event occurring to the apparatus.

In one embodiment, a computer system for analysing tachometer and vibration response data from an apparatus having one or more rotary components comprises:

data storage for storing vibration response data and tachometer data from the apparatus, the tachometer data being for a rotary component of the apparatus, and the vibration response data and tachometer data being acquired independently of each other, and

a processor for (a) determining a forcing frequency of the component from the tachometer data and a corresponding vibration response frequency of the apparatus from the vibration response data, and (b) comparing the forcing and vibration response frequencies to determine the relative phase difference between the frequencies.

The present inventors have also realised that polar diagrams which represent phase difference and vibration amplitude measurements can be used in novel ways to model engine behaviour and monitor for engine abnormalities.

Thus a second aspect of the invention provides a method of constructing a model of normal behaviour for an apparatus having at least one rotary component, the method comprising the steps of:

measuring, for a rotational speed of the component, (a) the phase difference between a forcing frequency for the component and a corresponding vibration response frequency of the apparatus, and (b) the vibration amplitude of the response frequency; and

using the measured phase difference and corresponding vibration amplitude to determine, for the rotational speed, a perimeter which sets the limit of normal behaviour for that speed, the perimeter being plotted on a polar diagram which represents phase differences and vibration amplitudes.

Preferably the centre of the area on the polar diagram encompassed by the perimeter is the average position of successive phase difference and vibration amplitude

measurements at the rotational speed. The perimeter radius may scale with the amount of scatter between the positions of the successive phase difference and vibration amplitude measurements. In this way, the method allows a statistically-based model of normal behaviour to be determined in situations where only intermittent operation occurs at each rotational speed. For example, an aero gas turbine engine under normal service conditions does not usually undergo smooth acceleration and deceleration manoeuvres between 0% and 100% speed, but rather spends varying amounts of time at different speeds. However, by applying the method of this aspect of the invention to each of a plurality of speeds, which preferably together cover the full range of speeds, a model of normal in service behaviour across the full range can be gradually accumulated without the need for dedicated acceleration and deceleration manoeuvres. A model of this type can be particularly useful for identifying slowly evolving engine abnormalities.

Alternatively, the model may be continually updated during operation of the apparatus. The centre of the area on the polar diagram encompassed by the perimeter can then be the position of the current phase difference and vibration amplitude measurement. A perimeter radius may then be set which scales with e.g. the amount of noise in the measurements. If the position of the next phase difference and vibration amplitude measurements are outside the perimeter, an engine abnormality may be the cause. Of course intentional changes in speed may also cause the position of the next phase difference and vibration amplitude measurements to move, but when such changes occur the perimeter radius can be momentarily expanded to compensate. This type of model can be particularly useful for identifying relatively quickly occurring engine abnormalities.

Preferably the apparatus is a gas turbine engine. The rotary component may be a turbine drive shaft of the engine.

Preferably the method of the first aspect of the invention is used to supply the measurements of phase difference and vibration amplitude. However, this is not essential and, for example, vibration response data may be acquired under the control of a tachometer (e.g. according to Figure 1) and then used in the measurement of the phase difference and vibration amplitude.

Related aspects of the present invention provide computer readable media carrying a model of normal behaviour constructed according to the second aspect, and the use of such a model for monitoring for abnormal behaviour in an apparatus having at least one rotary component.

By "computer system" we mean the hardware, software and data storage devices used to perform the method of a previous aspect. For example, a computer system of the present invention may comprise a central processing unit (CPU), input means, output means and data storage. Desirably the computer system has a monitor to provide a visual output display e.g. for polar diagrams. The data storage may comprise RAM or other computer readable media.

By "computer readable media" we mean any medium or media which can be read and accessed directly by a computer or computer system. The media include, but are not limited to: magnetic storage media such as floppy discs, hard disc storage medium and magnetic tape; optical storage media such as optical discs or CD-ROM; electrical storage media such as RAM and ROM; and hybrids of these categories such as magnetic/optical storage media.

Brief Description of the Drawings

The various aspects of the invention will be further described by way of example with reference to the accompanying drawings, in which:

Figure 1 is a flow diagram showing the sequence of data acquisition and analysis events in a conventional approach for determining the phase relationship between a forcing frequency and a vibration response;

Figure 2 shows a schematic longitudinal section of the coaxial shafts of a Rolls-Royce multi-shaft gas turbine engine;

Figure 3 shows schematically a vibration amplitude plot for a typical vibration response to an engine order forcing frequency during engine acceleration;

Figure 4 shows schematically the signals received from tachometer 5 of Figure 2;

Figure 5 is an example of a transform window;

Figure 6 is a flow diagram showing the sequence of data acquisition and analysis events in an approach for determining the phase relationship between an engine forcing frequency and a vibration response according to an embodiment of the present invention;

Figure 7a shows schematically a polar diagram for the same engine acceleration as Figure 3; and Figure 7b shows schematically a further polar diagram for a different acceleration by the same engine;

Figures 8a and b show respectively corresponding tracked orders and polar diagram plots for real data collected from a decelerating Rolls-Royce Trent engine;

Figure 9a shows four schematic plots, labelled 14a-d, of shaft vibration amplitude for respective "mass redistribution" events in a gas turbine engine; and Figure 9b shows four corresponding schematic polar diagram plots, labelled 15a-d, for the same events as Figure 9a;

Figures 10a and b show respectively a vibration amplitude plot and the corresponding polar diagram for real data collected from a Rolls-Royce Trent engine operating at steady state which experienced a blade detachment event;

Figures 11a and b show respectively a schematic LP vibration amplitude plot and the corresponding polar diagram for a gas turbine engine experiencing a bird strike;

Figures 12a and b show respectively a vibration amplitude plot and the corresponding polar diagram for real data collected from a Rolls-Royce Trent engine operating at steady state which experienced an actual bird strike;

Figure 13a shows a polar diagram plot for a typical acceleration-deceleration manoeuvre superimposed with three circles corresponding to average position and 3σ limits for respective speed sub-ranges determined from 100 flying hours of data; and Figure 13b is a detail of the high speed end of the plot of Figure 13a.

Figures 14a and b show schematically an HP vibration amplitude plot and the corresponding polar diagram for a Rolls-Royce Trent gas turbine engine experiencing progressive bearing failure; and

Figures 15a and b show schematically a vibration amplitude plot and the corresponding polar diagram for a shaft of an accelerating and subsequently cruising gas turbine engine, one of the shaft fan blades having a growing skin crack.

Description of the Embodiments

Figure 2 shows a schematic longitudinal section of the three coaxial shafts of a Rolls-Royce multi-shaft gas turbine engine. Low pressure (LP) shaft 1 occupies the central position and is surrounded in turn by intermediate pressure (IP) shaft 2 and high pressure (HP) shaft 3.

LP shaft 1 carries a single pole tachometer 4 which is conventionally used for shaft balancing operations. IP shaft 2 carries a 60 pole tachometer 5 which is conventionally used for standard shaft speed measurements for engine control purposes. HP shaft 3 is connected via bevel gear 6 and step aside gearbox 7 to main (external) gear box 8, with a tachometer 9 connected to the output of the main gear box.

Tachometer 4 produces a signal pulse for each rotation of the LP shaft, and hence the rotational frequency (i.e. shaft speed) of the LP shaft can be determined from the inverse of the period between signal pulses. Tachometer 5 produces 60 signal pulses for each rotation of the IP shaft, and hence the rotational frequency of the IP shaft can be determined from the inverse of 60× the period between signal pulses. Tachometer 9 measures the rotational frequency of the output of the main gearbox. However, as the gearing ratios of the main and step aside gearboxes 7, 8 and bevel gear 6 are known, the rotational frequency of the HP shaft can be inferred from tachometer 9. The rotational frequencies measured by the tachometers provide the forcing frequencies (e.g. fundamental, fractional and harmonic components) for each shaft.

Tachometer 4 also provides an absolute measure of shaft rotational position, whereas tachometers 5 and 9 provide relative measures of shaft rotational position.

A vibration transducer (not shown in Figure 2) is fitted to the engine and independently acquires vibration response data.

The vibration response data is sampled at an appropriate rate for the bandwidth of interest and is processed by an FFT in the conventional manner, retaining real and imaginary components. The phase derived for each real and imaginary pair is referenced to the phase angle of the first frequency bin in the transform window.

The vibration amplitude against time of a typical vibration response to an engine order forcing frequency (measured by tachometer 4, 5 or 9) during engine acceleration is shown schematically in Figure 3. As the engine speed (and hence forcing frequency) increases, the amplitude of the corresponding vibration response varies. Such plots are herein termed "vibration amplitude plots"

The tachometer signals are sampled at higher frequencies than the vibration response data. The shaft rotational frequencies (and hence fundamental forcing frequencies) are conveniently calculated by locating the positive rising transitions of the signal pulses. This is illustrated in Figure 4, which shows schematically the signals received from tachometer 5. Each pulse 20 is caused by the passage of one of the 60 poles. Positive rising transitions are identified by arrows.

Phase information from each tachometer signal is determined by locating the positive rising transition nearest to the centre of the transform window. The distance of this transition from the centre represents a time delay, t, between the transition of the tachometer signal and the response signal of the shaft. The phase of the shaft (and the respective fundamental forcing frequency) can be derived from the expression $2\pi ft$, where f is the rotational frequency of the shaft calculated previously.

Figure 5 is an example of a transform window showing the vibration signal for the first frequency bin 21, the tachometer signal 20, and time delay t. The window centre is indicated by the dotted vertical line.

In this way, each vibration response phase measurement is compared with the forcing frequency phase measurement for the same instant to derive a relative phase difference between the forcing frequency and the vibration response frequency. Essentially the forcing and vibration response frequencies are compared in the time domain. Fractional and harmonic frequencies and phases can be calculated from the fundamental forcing frequency and compared with the vibration response frequency in the same way. Relative phase differences can also be calculated for non-integral order related components (such as the radial drive shaft, gears, pumps etc.) which are directly or indirectly connected to the main shafts and have a fixed relationship of rotational speed to the monitored tachometer signal.

Figure 6 is a flow diagram showing the sequence of data acquisition and analysis events according to this embodiment of the present invention. In contrast with Figure 1, note how the data acquisition steps are independent of each other.

Figure 7a shows schematically a polar or Nyquist diagram which plots the vibration amplitude of Figure 3, and the relative phase difference between the vibration response of Figure 3 and the engine order forcing frequency which produced the response. The vibration amplitude is represented as distance from the origin and the phase difference as angular position. The overall direction of increasing engine speed is indicated by the arrow. Such diagrams are herein termed "polar diagrams".

The plot follows a characteristic path as the engine accelerates. In particular, the relative phase difference between the vibration response and the forcing frequency changes as the engine speed increases. These changes are primarily caused by the traversal through engine resonance frequencies. The overall looped shape of the path is caused by engine resonances. The maxima in Figure 3 can now be seen to be caused by the resonances. When the engine reaches its cruising speed the relative phase difference and vibration amplitude do not vary significantly, and the plot tends to remain within the relatively tightly demarcated area indicated by the dashed circle of Figure 7a.

If the engine performs a series of successive acceleration/deceleration manoeuvres, paths having the same characteristic shape are formed on the polar diagram. However, in cases where the engine is shut down between manoeuvres, the measured relative phase difference between the vibration response and engine order forcing frequencies may be shifted. This shift manifests itself in the polar diagram as a rotation of the path about the origin. Figure 7b shows schematically a polar diagram for the same forcing frequency as Figure 7a, but rotated in the manner described.

Figures 8a and b show respectively tracked order and polar diagram plots for real data collected from a decelerating Rolls-Royce Trent engine. The fundamental tracked orders and corresponding plots on the polar diagram are respectively labelled LP, IP and HP. Although the polar diagram plots are "noisier" than the schematic diagrams of Figure 7, the characteristics features of the diagrams can be identified, particularly on the LP plot.

Such polar diagrams are particularly useful for providing indications of abnormal or unhealthy engine behaviour. In

particular, a deviation from normal engine behaviour may manifest itself as a variation in relative phase difference and/or vibration amplitude, and this in turn can be readily identified by a departure from the characteristic path of a polar diagram.

Figure 9a shows four schematic plots, labelled 14a-d, of shaft vibration amplitude for respective "mass redistribution" events (in this case a detached blade) in a gas turbine engine. Before and after the events the engine cruises at a constant speed. In a conventional detection system, an alert may be signalled if the vibration amplitude exceeds a threshold level, which in Figure 9a is indicated by a dashed line. However, of the four possible scenarios shown in Figure 9a, only one produces a significantly high increase in vibration amplitude to signal an alert, although potentially all the events are equally noteworthy.

Figure 9b shows four corresponding schematic polar diagram plots, labelled 15a-d, for the same events as Figure 9a. Before the events all the plots are clustered within the smaller dashed circle in the same part of the diagram. After the events each plot relocates to a position on the larger dashed circle which represents possible outcome scenarios.

Note how even events, such as that represented by plot 15b, which produce no change in vibration amplitude are easily identified on the polar diagram (the corresponding plot on Figure 9a is 14b). Also events, such as that represented by plot 15a, which produce a reduction in vibration amplitude and so give the appearance of a more smoothly running engine (the corresponding plot on Figure 9a is 14a), can be readily identified as engine abnormalities.

Figures 10a and b show respectively vibration amplitude and polar diagram plots for real data collected from a Rolls-Royce Trent engine operating at steady state which experienced an actual blade detachment event. The vibration amplitudes and corresponding relative phase difference are for the fundamental shaft tracked orders and are respectively labelled LP, IP and HP. In this case the blade detached from the HP shaft, which did in fact respond with a sharp increase in vibration. Note, however, how the LP and IP vibration amplitudes varied only slightly, whereas their relative phase differences shifted significantly.

Thus polar diagrams may be used to provide warning systems for engine malfunction or abnormal behaviour. An example of such a system for detecting step changes during what should be steady state operation may be implemented with the following pseudo code:

If engine is in steady state (e.g. <0.3% speed change in last 2 sec):

1) Record present position r on polar diagram

2) Update estimate for noise radius (i.e. error in present position) based on at least ten previous position measurements

3) Calculate step change score:

$$\text{score} = \frac{\Delta r}{\text{noise radius}} - (k.\Delta s + C)$$

where Δr is the distance on the polar diagram between the present and the previous position, Δs is the change in speed, and $(k.\Delta s + C)$ is a linear relationship between shaft speed and change in phase plot position (k and C being empirically derived constants)

4) If the step change score exceeds a predetermined threshold, declare an abnormal behaviour.

The system continuously updates the estimated variation due to noise in the vibration response data. Effectively, the system

draws a circle on the diagram within which the plot of vibration amplitude and relative phase difference can wander randomly at a given engine speed. This circle defines a perimeter outside of which abnormal behaviour is declared. Since vibration amplitude and relative phase difference are expected to vary as the engine speed changes, the allowed changes to the plot are scaled by the rate of change of speed. The overall effect is for the radius of the circle to increase during changes of speed.

A further example of an event that could be detected by such a system is a step change caused by foreign object damage. Figures 11a and b respectively show a schematic LP vibration amplitude plot and the corresponding polar diagram for a foreign object damage event (in this case a bird strike) to a gas turbine engine. The bird strike produces only a small temporary deviation (of about 0.5 s duration) on the vibration amplitude plot, which could easily be overlooked or missed altogether. On the polar diagram, however, the bird strike provokes a readily detectable and significant change in relative phase difference.

Figures 12a and b show respectively vibration amplitude and polar diagram plots for real data collected from a Rolls-Royce Trent engine operating at steady state which experienced an actual bird strike. As before, the vibration amplitudes and corresponding relative phase difference are for the fundamental shaft tracked orders and are respectively labelled LP, IP and HP. As might be expected, the vibration amplitude for the LP shaft increased sharply. Note, however, how the IP shaft vibration amplitude actually decreased after the strike, and the HP shaft vibration amplitude behaviour was largely unchanged. In contrast, the relative phase differences for the three shafts were all disturbed by the bird strike, as shown by the polar diagram.

The examples of Figures 9 to 12 show how the relative phase difference between forcing and vibration response frequencies, and its representation on a polar diagram, can be used to detect events which lead to step changes during what should be steady state operation. A similar approach is also useful, however, for detecting and monitoring more slowly evolving engine abnormalities.

As a first step an appropriate model of normality against which slowly evolving engine changes can be tracked is established. In service, aero engines do not usually experience the acceleration/deceleration manoeuvres which produced the polar diagram plots shown in previous Figures. Instead they spend differing amounts of times at different engine speeds. Thus, for example, if maximum engine speed is designated 100%, most engines spend much of their operational time in the 50-70% speed range.

In order to build up the normal model, the entire speed range is divided into a smaller (e.g. 1%) sub-ranges. Each time the engine passes through a sub-range, the relative phase difference and vibration amplitude are recorded. As more data is acquired over further flights an average position (i.e. phase difference and vibration amplitude) on the polar diagram and a standard deviation from that position is associated with each sub-range. It may require e.g. 100 flying hours or 25 flight cycles to provide enough data to accurately characterise the entire speed range in this way. As the engine may be shut down between flights, the measured relative phase differences may have to adjusted to account for phase shifts between the vibration response and engine order forcing frequency. This can be accomplished by using one or more well-characterised speed sub-ranges (e.g. in the 50-70% range) as a reference.

Figure 13a shows a polar diagram plot for a typical acceleration-deceleration manoeuvre. Superimposed on the plot are three circles 30. The centre of each circle is the average measured position for a particular speed sub-range determined from 100 flying hours of data. The circumference of each circle is at the 3σ (3 × the standard deviation) radius for the measured position data. Each circle, therefore, is a perimeter setting the limit of normal behaviour for respective speed sub-range.

Figure 13b is a detail of the high speed end of the plot of Figure 13a. The average positions for the speed sub-ranges centred on 97% and 98% speed respectively are indicated by dots 31a and 31b and the corresponding 3σ circles 30a and 30b are also shown. The progression of dots 32 are real-time average positions for 98% speed, the arrow indicating the time order in which the average positions were measured. The observed evolution of 98% speed average position is characteristic of a progressive failure mode. This failure is flagged when the 98% speed average position oversteps the 3σ perimeter.

The following examples illustrate the detection of progressive failure modes using this approach. Figure 14a shows schematically an HP vibration amplitude plot for a Rolls-Royce Trent gas turbine engine. At time A the engine has just reached cruising speed. There follows an approximately four minute period in which the HP vibration amplitude decreases to almost zero. A superficial interpretation of this decrease might be that the engine is running more smoothly. However, in the six subsequent minutes, the vibration amplitude increases until at time B an HP location bearing failure occurs. In retrospect, therefore, it is clear that the decrease and increase in vibration amplitude was actually caused by a progressive bearing failure.

Figure 14b shows the corresponding HP polar diagram. Up to time A, the path of the polar diagram follows that of the normal model. Then, rather than remaining approximately stationary when cruising speed is attained, the path veers off in a straight line. Thus from the polar diagram the departure from normality is apparent as soon as the 3σ limit for cruising speed is overstepped (i.e. almost immediately), allowing the engine operator to take remedial action at an earlier stage. Note that the point of closest approach of the straight line to the origin corresponds to the minimum in the HP vibration amplitude shown in Figure 14a.

Figures 15a and b again show schematically a vibration amplitude plot and the corresponding polar diagram for an accelerating and subsequently cruising gas turbine engine. In this case a fan blade having a relatively well-developed skin crack was planted in the engine. The crack did not grow significantly during engine acceleration, and the vibration amplitude plot and polar diagram followed their normal paths as the engine accelerated to cruising speed at time A. However, over a period at cruising speed the crack grew until the blade failed. As there was little variation in the amplitude of the vibration response during this period, the vibration amplitude plot gave little warning of the incipient failure. In contrast, the growing crack produced a significant change in relative phase difference which was immediately identifiable from the polar diagram.

Other gas turbine engine problems which can be identified from variations relative phase difference behaviour and/or polar diagrams include unbalance, squeeze film bearing problems, rubs, joint movements, oil in drum, ½ speed whirl, instability and blade scatter. Indeed, in certain cases it may be possible to diagnose the cause of the problem from the form of the polar diagram. For example, a typical response to rub is

an increase in response vibration amplitude at constant relative phase difference.

CLAIMS

1. A method of analysing tachometer and vibration response data from an apparatus having one or more rotary components, the method comprising the steps of:

providing vibration response data and tachometer data from the apparatus, the tachometer data being for a rotary component of the apparatus, and the vibration response data and tachometer data being acquired independently of each other;

determining a forcing frequency of the component from the tachometer data and a corresponding vibration response frequency of the apparatus from the vibration response data; and

comparing the forcing and vibration response frequencies to determine the relative phase difference between the frequencies.

2. A method according to claim 1, wherein the apparatus is a gas turbine engine.

3. A method according to claim 2, wherein the rotary component is a turbine drive shaft of the engine.

4. A method according to any one of the previous claims, wherein the tachometer data does not provide the absolute rotary position of the component.

5. A method according to any one of the previous claims, wherein the forcing and vibration response frequencies are compared in the time domain in order to determine the relative phase difference between the frequencies.

6. A method according to any one of the previous claims, wherein the amplitude of the vibration response is further determined from the vibration response data.

7. A method according to claim 6, further comprising the step of:

plotting the relative phase difference and the vibration amplitude on a polar diagram.

8. A method according to any one of claims 1 to 7, further comprising the step of:

using the relative phase difference to diagnose the state of the apparatus.

9. A method according to any one of claims 1 to 7, further comprising the step of:

using the relative phase difference to identify an event occurring to the apparatus.

10. A computer system operatively configured to perform the method of any one of claims 1 to 9.

11. Computer readable media carrying computer code for performing the method of any one of claims 1 to 9.

12. A computer program for performing the method of any one of claims 1 to 9.

13. A method of constructing a model of normal behaviour for an apparatus having at least one rotary component, the method comprising the steps of:

measuring, for at least one rotational speed of the component, (a) the phase difference between a forcing frequency for the component and a corresponding vibration response frequency of the apparatus, and (b) the vibration amplitude of the response frequency; and

using the measured phase difference and corresponding vibration amplitude to determine, for the rotational speed, a perimeter which sets the limit of normal behaviour for that

speed, the perimeter being plotted on a polar diagram which represents phase differences and vibration amplitudes.

14. Computer readable media carrying a model of normal behaviour constructed according to claim 13.

15. Use of a model constructed according to claim 13 for monitoring for abnormal behaviour an apparatus having at least one rotary component.

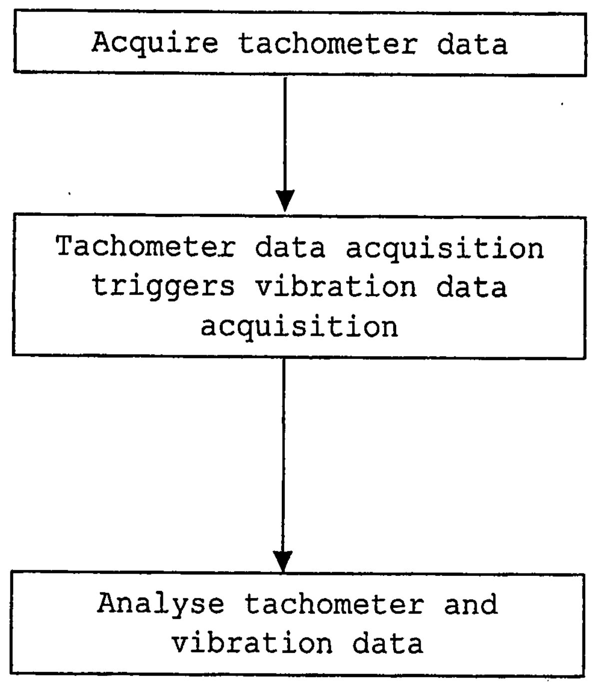

Acquire tachometer data
Tachometer data acquisition triggers vibration data acquisition
Analyse tachometer and vibration data

Figure 1

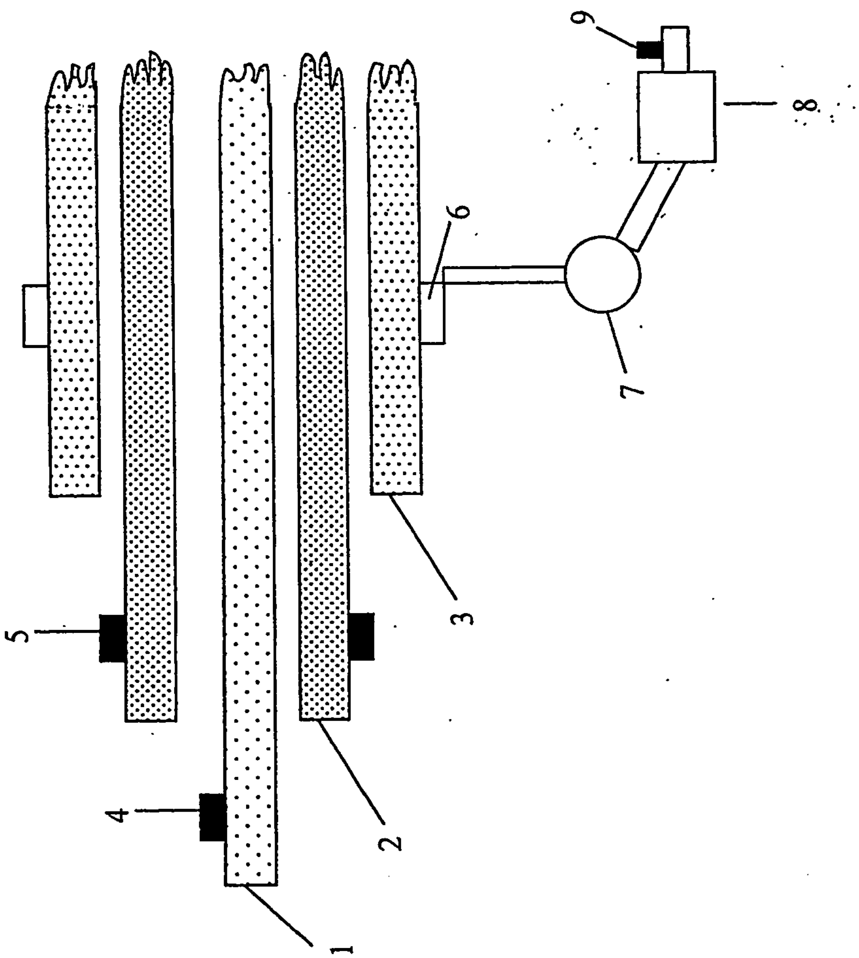


Figure 2

Figure 3

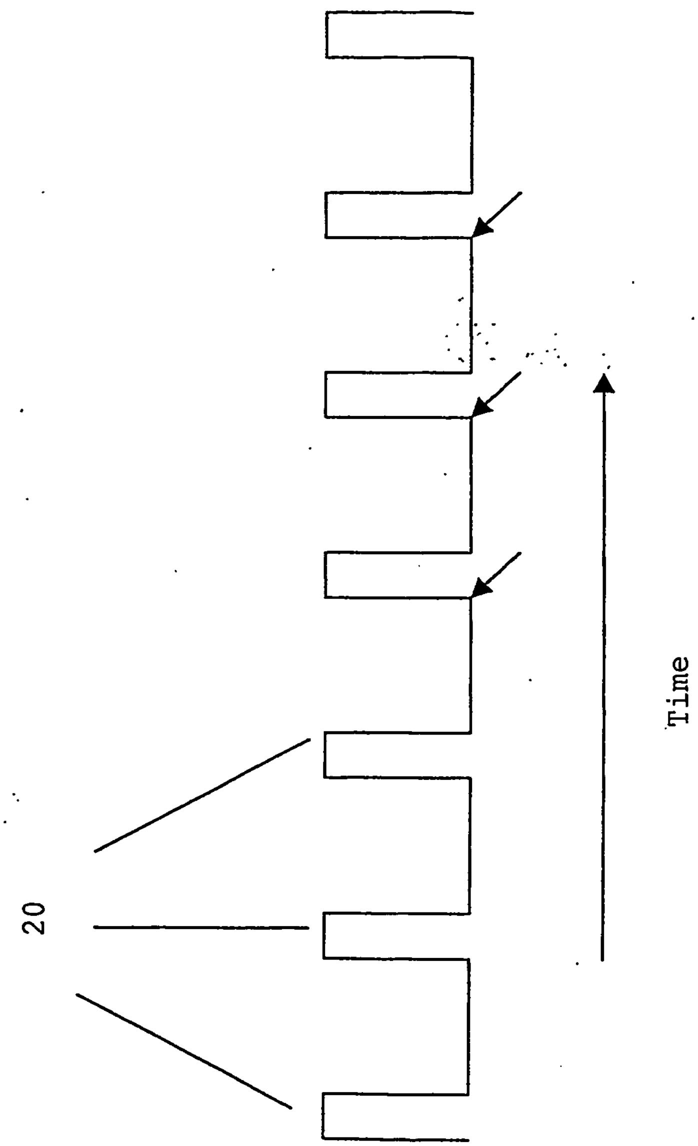


Figure 4

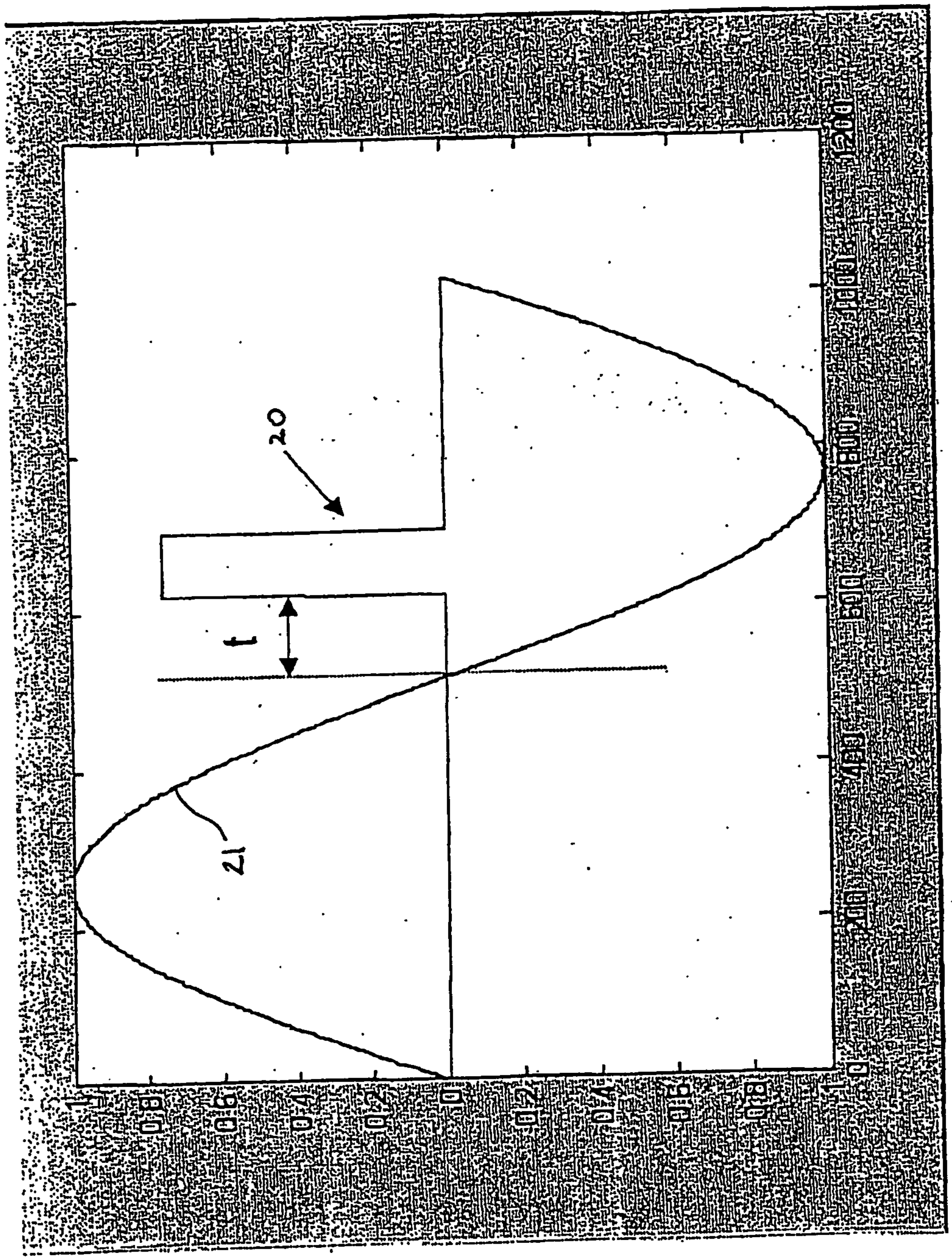


Figure 5

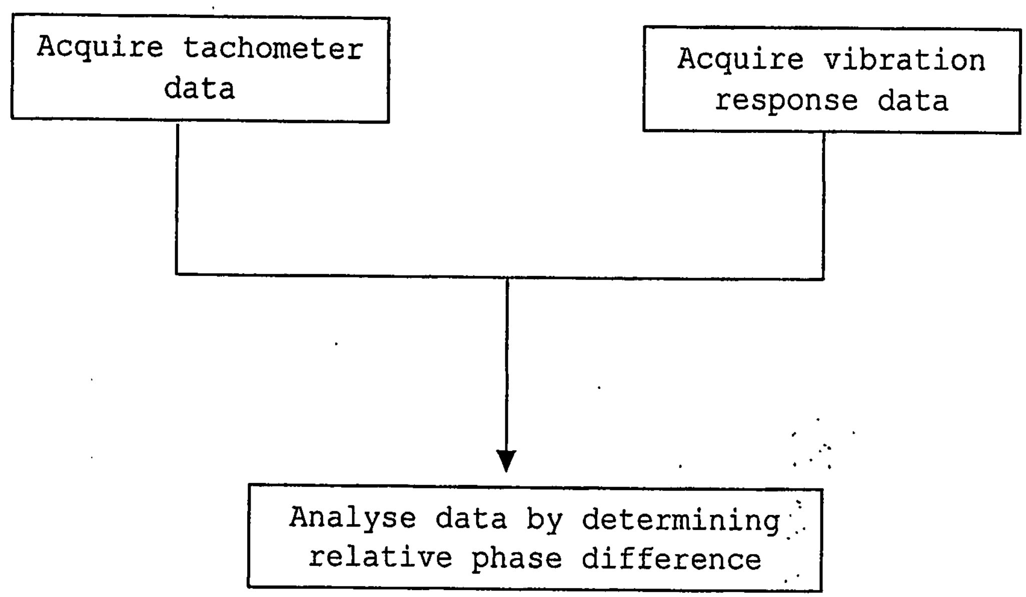

Acquire tachometer data
Acquire vibration response data
Analyse data by determining relative phase difference

Figure 6

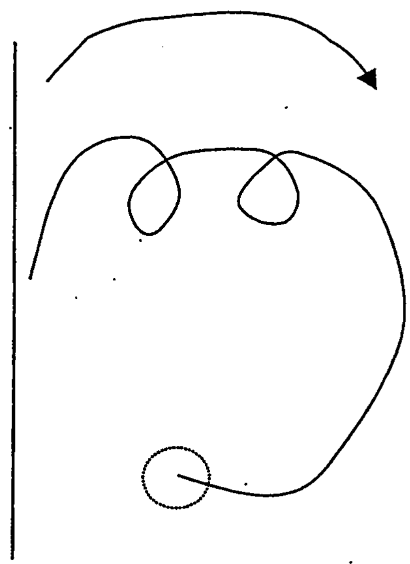


Figure 7a

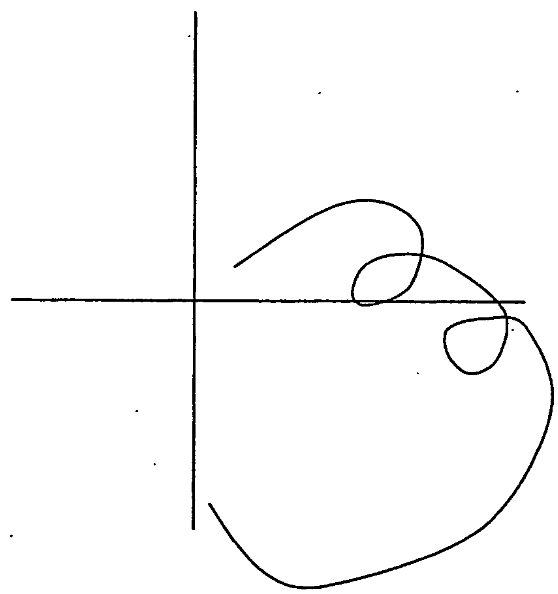


Figure 7b

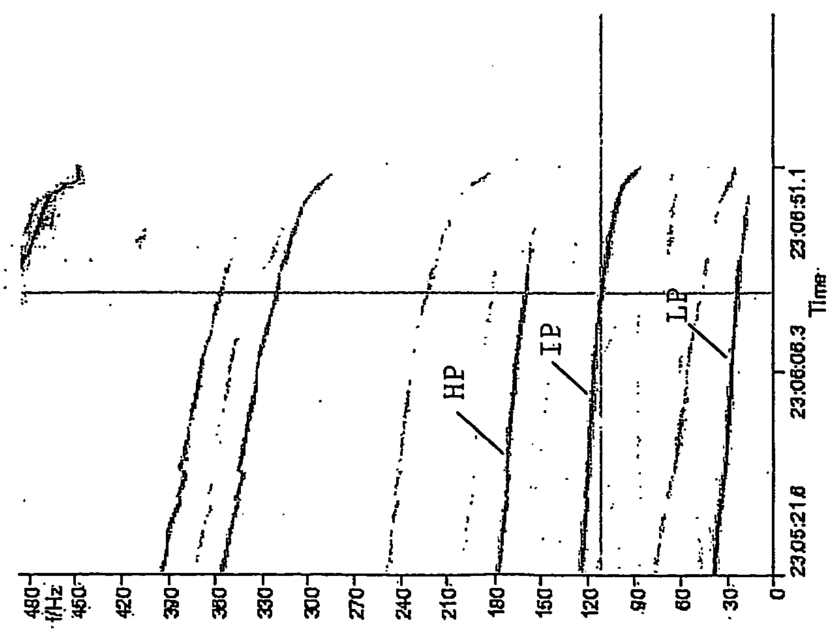


Figure 8a

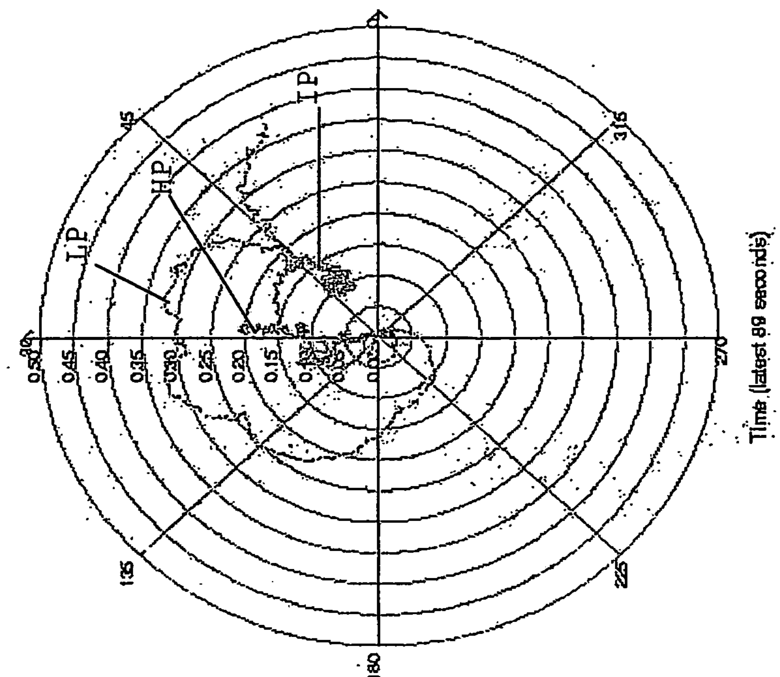


Figure 8b

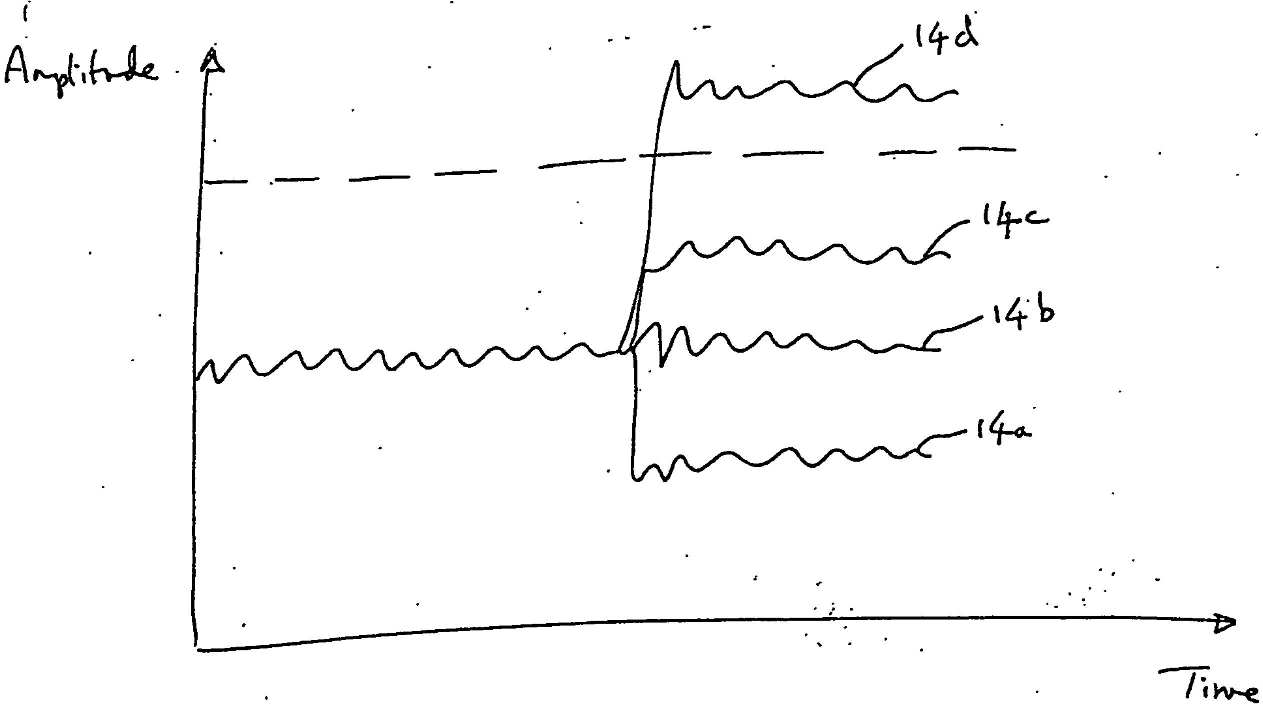


Figure 9a

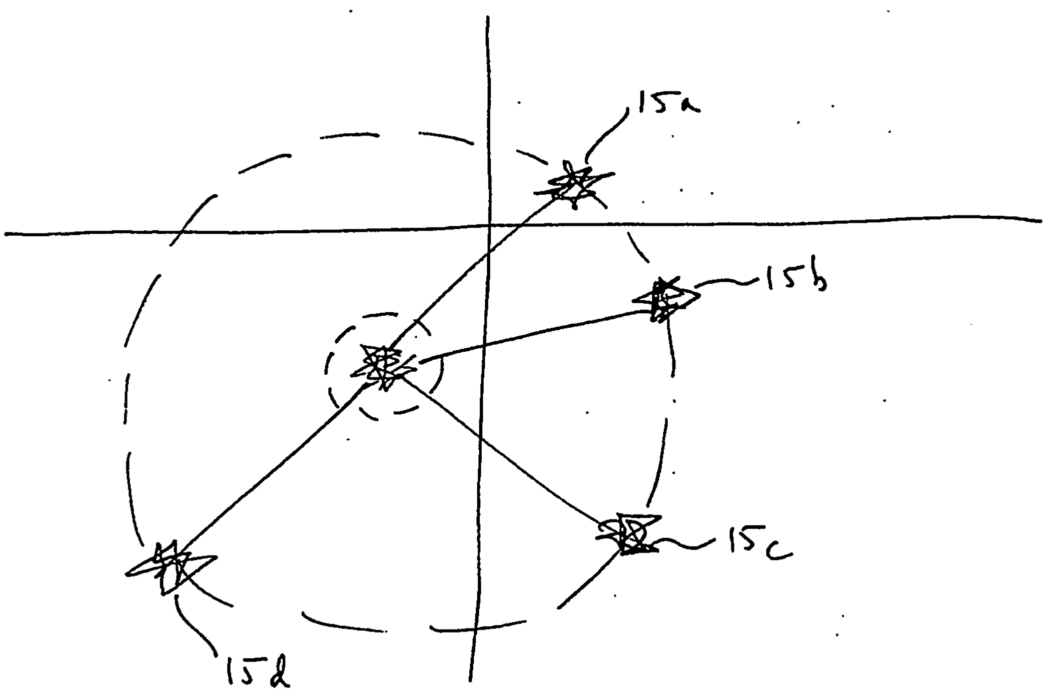
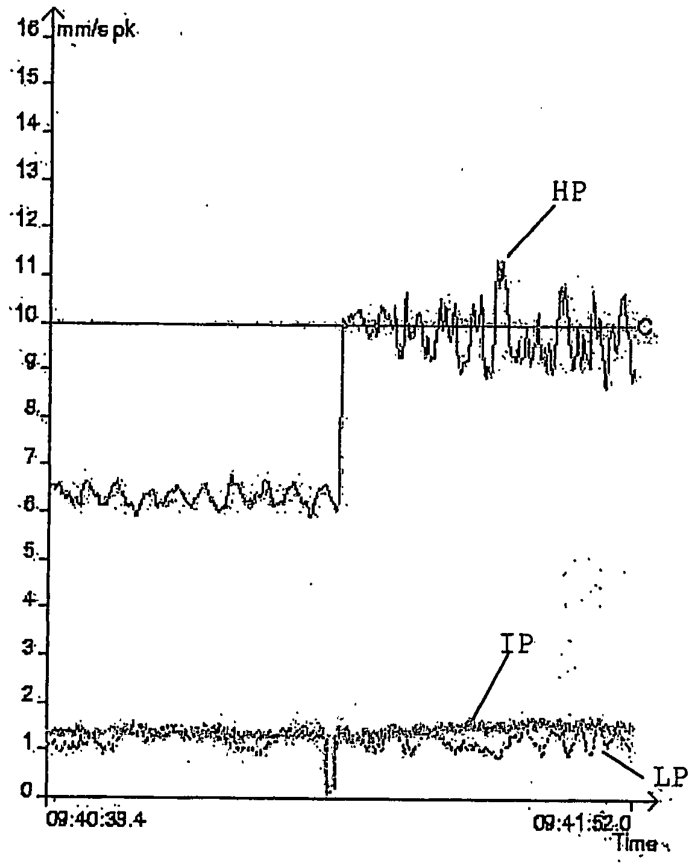


Figure 9b



mm/s pk
16
15
14
13
12
11
10
9
8
7
6
5
4
3
2
1
0
HP
C
IP
LP
09:40:38.4
09:41:52.0
Time

Figure 10a

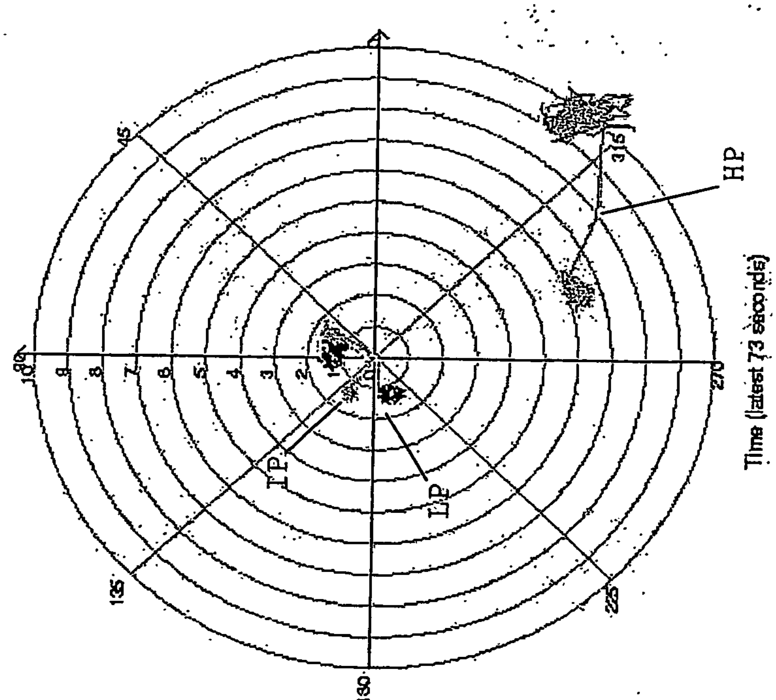

0
45
90
135
180
225
270
315
1
2
3
4
5
6
7
8
9
10
HP
LP
LP
Time (latest 73 seconds)

Figure 10b

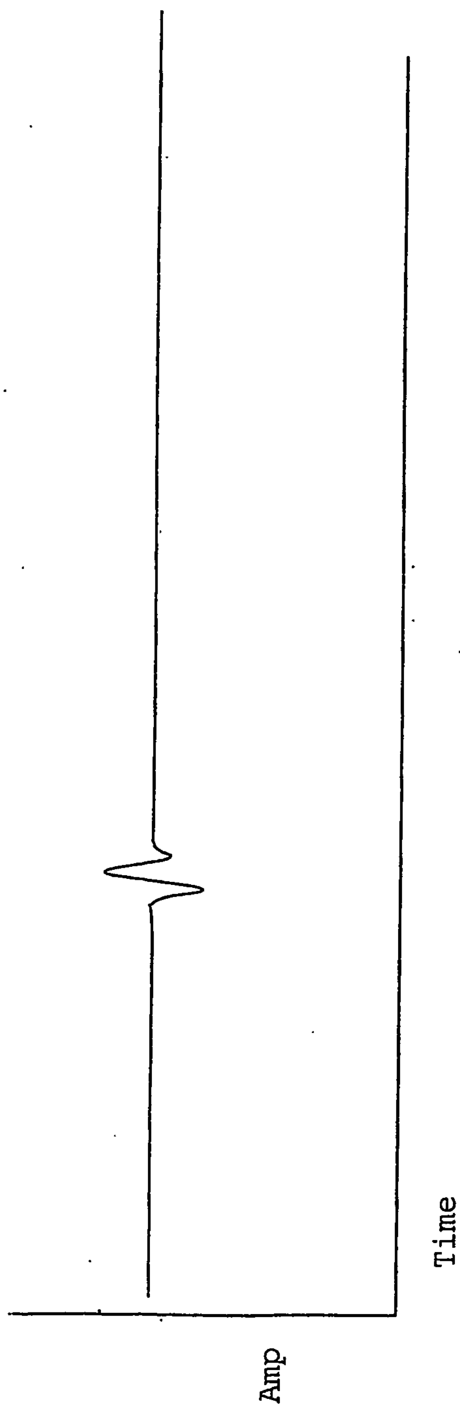


Figure 11a

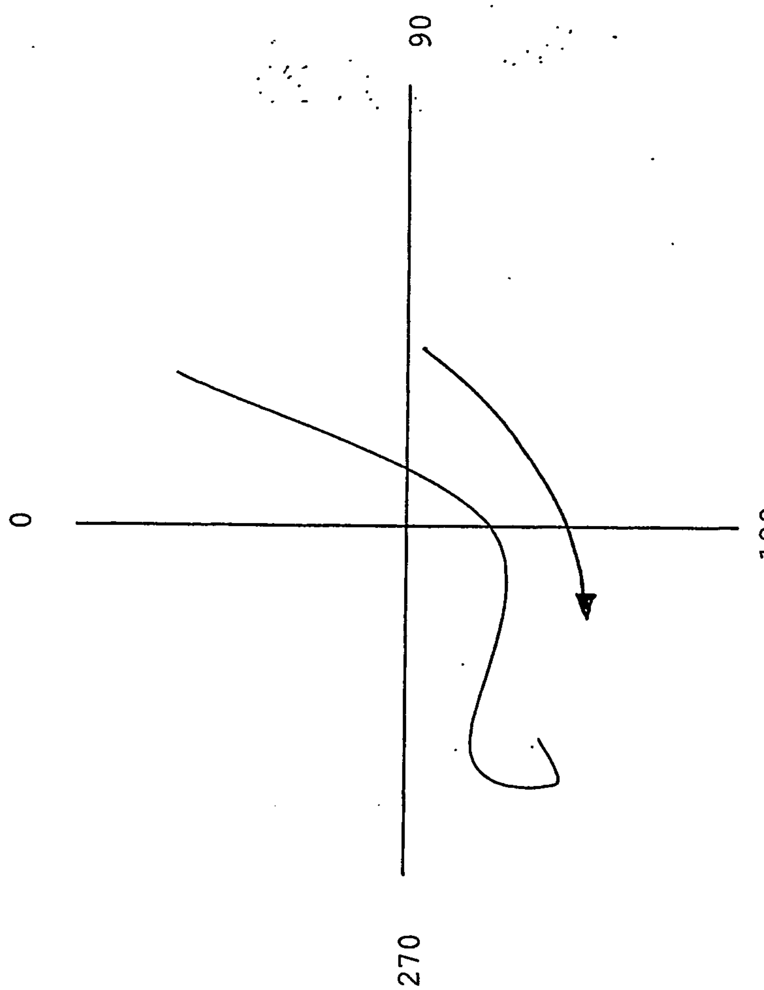


Figure 11b

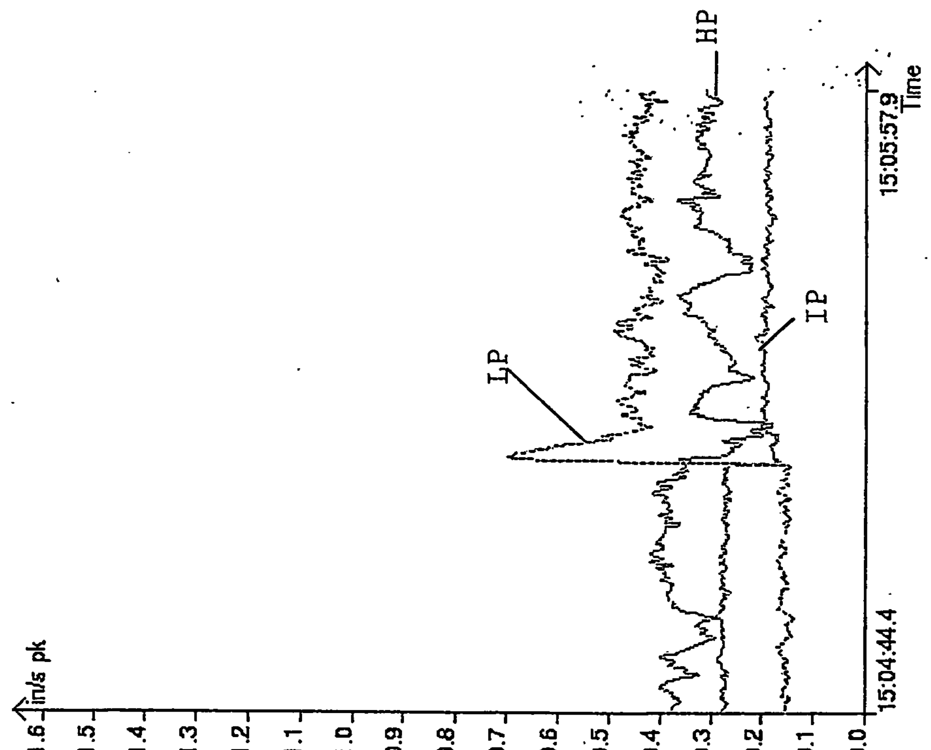

in/s pk
1.6
1.5
1.4
1.3
1.2
1.1
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0.0
15:04:44.4
15:05:57.9
Time
LP
IP
HP

Figure 12a

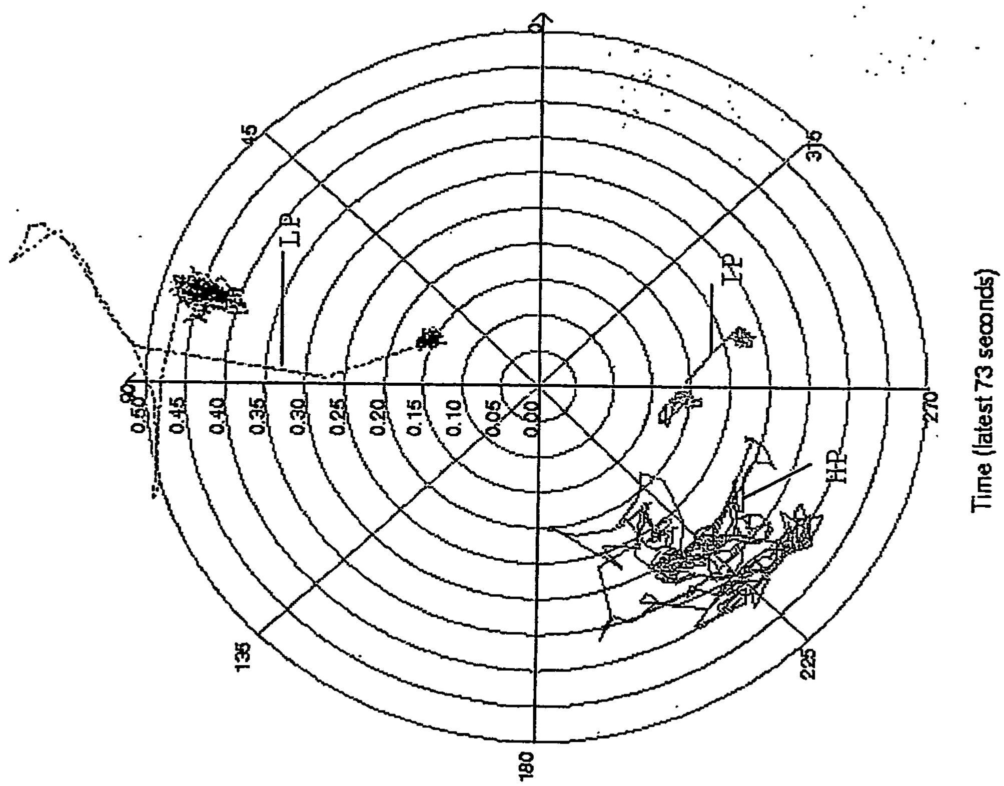

0
45
90
135
180
225
270
315
0.50
0.45
0.40
0.35
0.30
0.25
0.20
0.15
0.10
0.05
0.00
LP
LP
HP
Time (latest 73 seconds)

Figure 12b

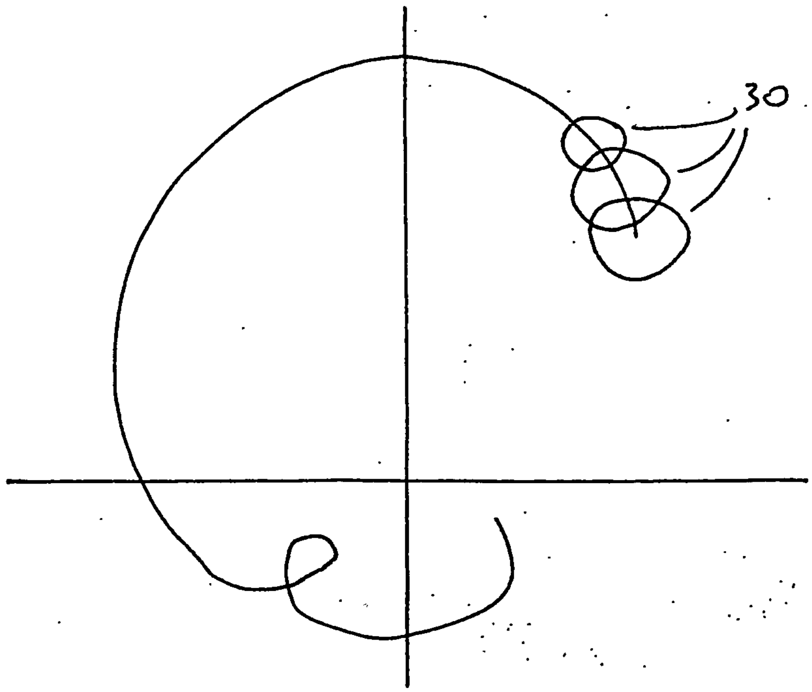


Figure 13a

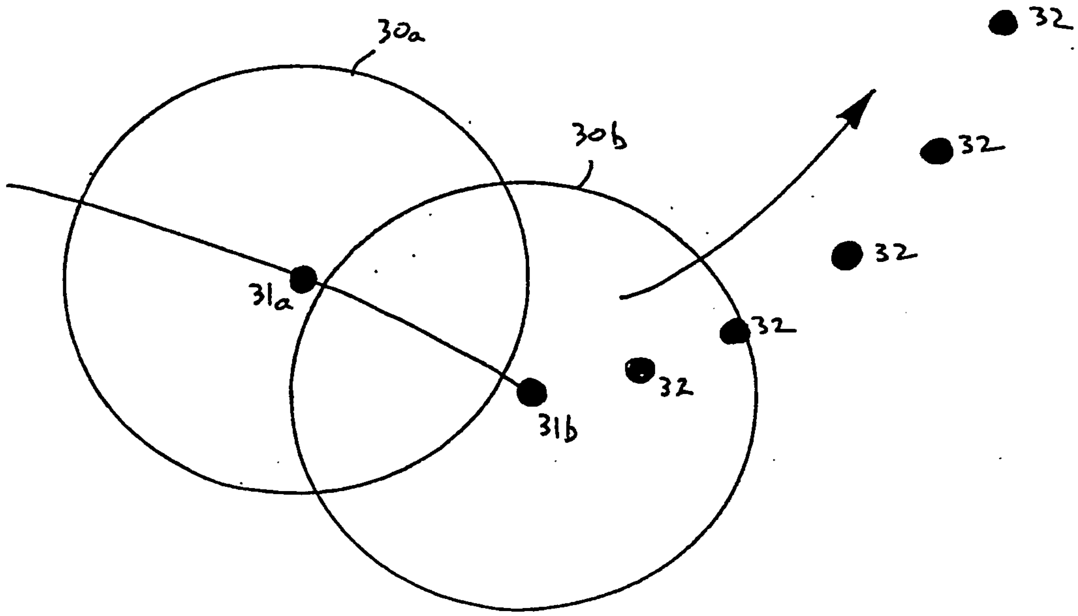


Figure 13b

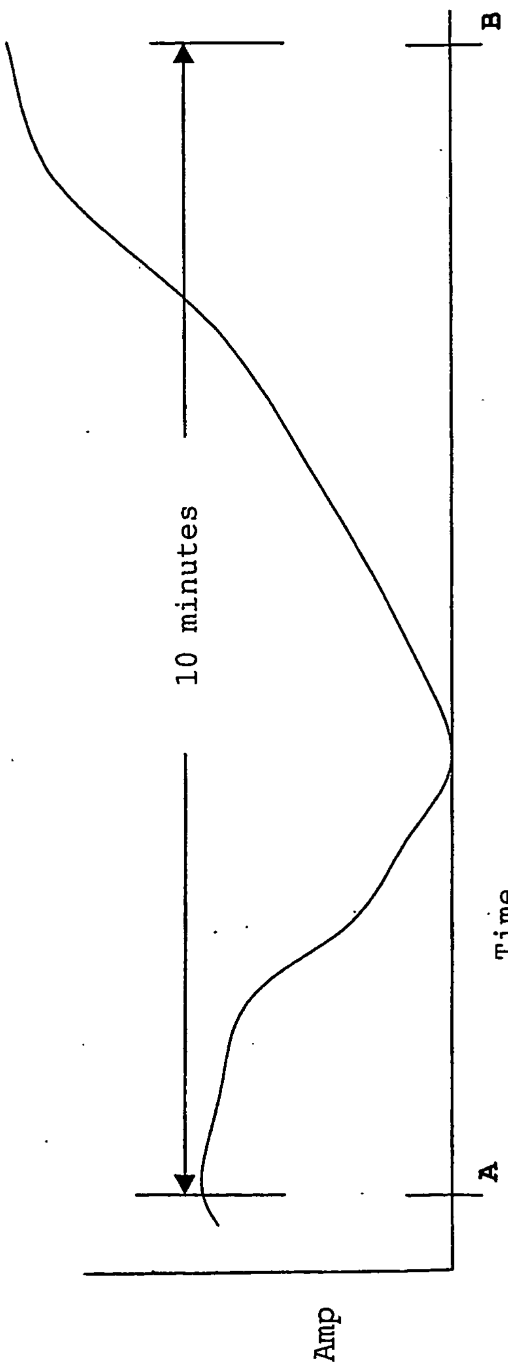


Figure 14a

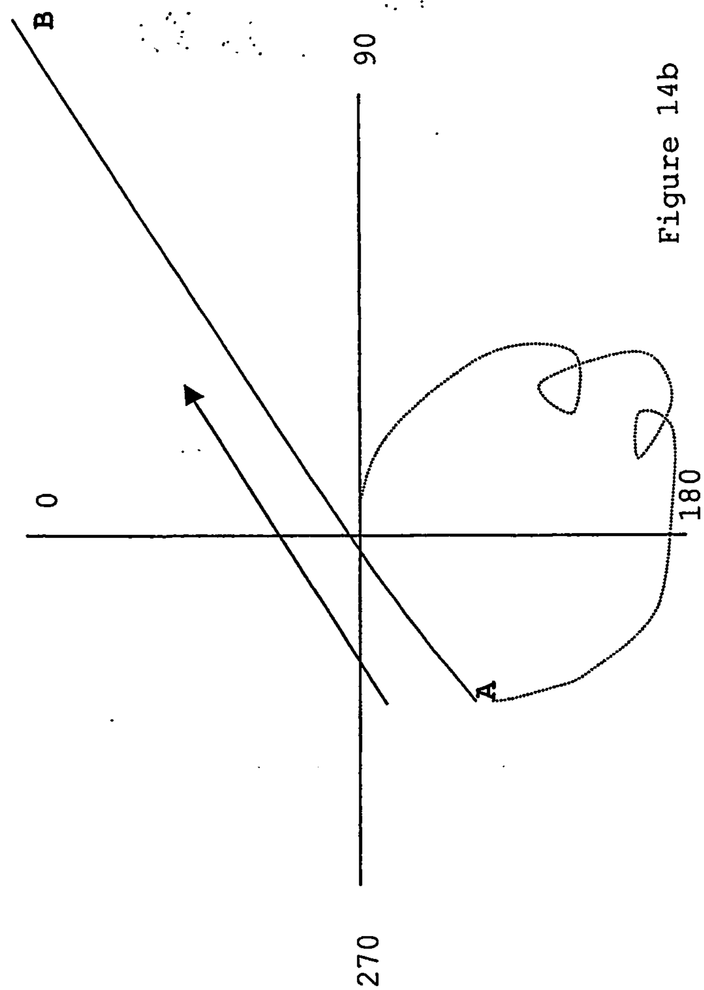


Figure 14b

Figure 15a

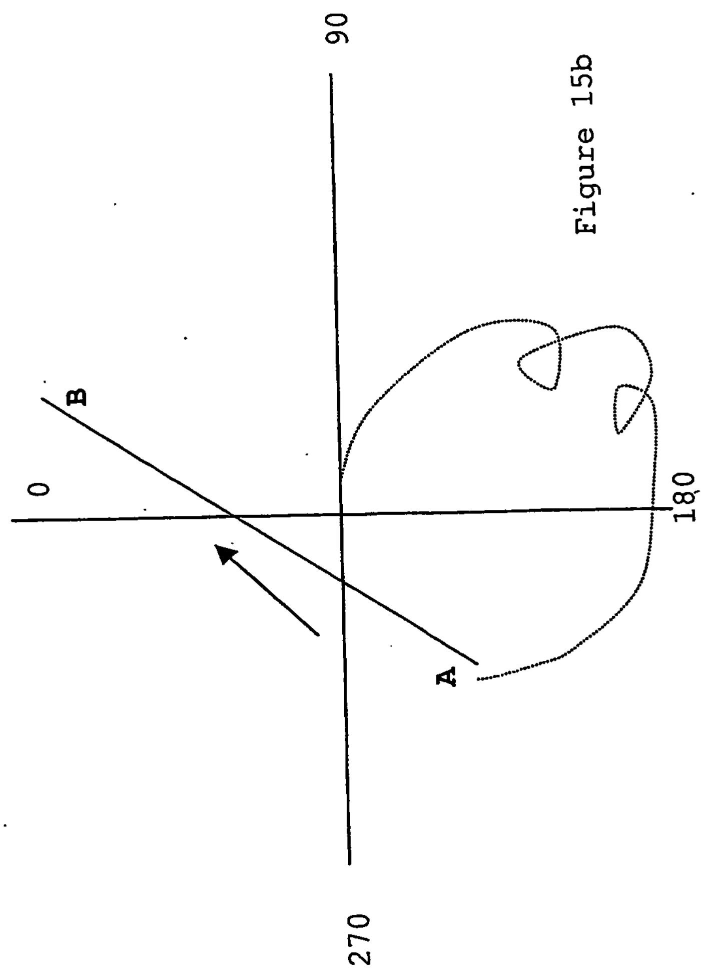


Figure 15b

PCT/GB2004/001575